AN APPLICATION OF THE
SERIES TURBINE
TO A
HIGH SPEED NAVAL COMBATANT SHIP

by

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AN APPLICATION OF THE SERIES TURBINE TO A HIGH SPEED NAVAL COMBATANT SHIP

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ABSTRACT

This thesis is an evaluation of the effects of modifying the machinery plant of the DLG-26 class of guided missile frigates by introducing a series turbine in the main steam line to drive the forced draft blowers and the fuel oil service pumps. Heat balances are recalculated and the resulting fuel rates are compared with those of the design heat balances; the automatic combustion control capability is investigated; reduced manning and remote control are discussed; and changes to the machinery arrangement are presented.

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Classified

LIST OF FIGURES

SYMBOLS

A	Area, in^2 ; constant
A.E.	Available energy, Btu/lb
BHP	Brake horsepower, hp
b	Impeller inlet width, in.
C	Velocity, absolute, ft/sec ; constant
C_p	Specific heat. Btu/lb-^oR
c	Turbine blade chord, in.
D	Velocity, relative, ft/sec.
D.L.	Draft loss, in. H_2O
d	Diameter, in.
FHP	Fan horsepower, hp
g	Gravitational Acceleration, ft/sec^2
H	Head, ft.
h	Enthalpy, Btu/lb
J	Constant, 778 ft-lb/Btu
K_B	Blade coefficient (dimensionless)
K_{co}	Carry-over loss coefficient (dimensionless)
K_i	Incidence loss coefficient (dimensionless)
K_n	Nozzle coefficient (dimensionless)
K_p	Profile loss coefficient (dimensionless)
K_{pw}	Profile loss coefficient, wet steam (dimensionless)
K'	Pressure coefficient (dimensionless)
k	Ratio C_p/C_v, (1.4 for air)
L	Height of blade, in.
LHP	Lost horsepower, hp

N	Number of nozzles; Revolutions per minute, rpm
n	Number of Labyrinths
O	Width of nozzle, in.
P	Nozzle passage width, in.; Pitch, in.; Power
p	Pressure, lb/in^2
PHP	Pump horsepower, h.p.
Q	Volume rate of flow (weight rate of flow in heat balances)
R	Universal gas constant
RPM	Revolutions per minute
r	Pressure ratio
SHP	Shaft horsepower, h.p.
T	Absolute temperature, ^{o}R
t	Temperature, ^{o}F
u	Tangential velocity, ft/sec.
U.E.	Used energy, Btu/lb.
V	Specific volume, $ft^3/lb.$
W	Weight rate of flow, lb/hr.
wk	Work, Btu/lb.
Y	Moisture, %
α	Nozzle exit angle, degrees
β	Blade entry angle, degrees
γ	Relative blade exit angle, degrees
Δ	Increment; labyrinth strip thickness, in.
δ	Radial clearance between shaft and labyrinth packing
ϵ	Admission fraction
η	Efficiency, %

θ	Deflection angle = $180-(\beta+\gamma)$, degrees.
λ	Blade exit angle - 90°, degrees
τ	Circumferential nozzle width
ϕ	Overall labyrinth pressure ratio factor

SUBSCRIPTS

A	Air
a	Actual
B	Bleed
c	Compression
F	Fuel
f	following
i	Ideal: initial
L	Loss
m	Mean
N	Nozzle
r	Radial
S	Steam
S.T.	Series Turbine
T	Turbine
x	Axial
'	Theoretical
o	Apparent: original
1,2,3 etc	Points along path of steam in turbine
1	Entering fan blade
2	Leaving fan blade

INTRODUCTION

The concept of the "series turbine" is one which may lead to a small revolution in the use of steam propulsion. Although a series turbine has yet to be installed in a ship, the theoretical advantages that are offered by the relatively simple revisions to the traditional steam plant hold out the promises of a major breakthrough in technology.

A series turbine is a small single stage turbine installed in the main steam line in series with the propulsion turbines. Its purpose is to drive the forced draft blowers and the fuel oil service pump which serve the boiler with which the turbine is associated. All of the steam which is generated by the boiler passes through the series turbine, and some energy is removed from the steam in driving the pump and blowers; however, the large amount of steam utilized results in a very small drop in the energy level of the steam as it passes through the series turbine. As visualized, the series turbine is placed in the main steam line between the steam drum and the superheater inlet, and so the energy that is removed by the series turbine can be replaced in the superheater.

The steam to the series turbine is not throttled,

even at low steaming rates, whereas the "parallel turbines", the auxiliary turbines which drive the forced draft blowers and fuel pumps of present steam plants, lose much in efficiency because the steam must be throttled to control the turbines' speeds. Since the steam consumption of the forced draft blowers is very high (almost 10% as much steam as is used by the propulsion turbines, at full power), the efficiency of the blower turbines is a significant factor in the fuel rate of the ship. Therefore the series turbine can significantly reduce the fuel rate which would otherwise result in a traditional steam plant.

A more efficient use of steam is only one asset of the series turbine. A characteristic of great importance is that the series turbine provides a complete automatic combustion control system without a single item of control equipment. As the demand for steam varies (by the throttleman's changing the throttle) the series turbine itself changes speed to provide the boiler with just the right proportions of fuel and air to generate the required amount of steam. It is a theoretically sound, reliable, uncomplicated combustion control system which requires no detailed training to operate and maintain.

A completely reliable automatic combustion control system is a pre-requisite for reduced manning and remote control. The series turbine's inherent self regulating capability permits reduced manning to the point of genuine remote monitoring, with absolutely no fireroom watch.

The theoretical advantages of the series turbine concept over present day steam machinery plants are of great potential value to the Navy. This thesis is an effort to evaluate the apparent benefits of the series turbine to determine if it has a real application in a Navy ship. The DLG-26 class of guided missile frigates was chosen to be modified in theory, and the objectives are to establish the fuel rates for the operating speeds from fifteen knots to full power for comparison with the fuel rates calculated in the original design heat balances; to rearrange the machinery and to revise plant parameters in order to give the most efficient plant; to determine if the series turbine arrangement would physically fit within the ship, utilizing the original boilers and turbines; and to show in theory how combustion control is accomplished and what the effect can be on manning and operation.

THEORY AND METHOD OF DEVELOPMENT

As was stated briefly in the introduction, a single stage turbine running unthrottled is more efficient than a throttled turbine. It can be demonstrated by examining the Mollier diagrams for a throttled and an unthrottled turbine of comparable power output. Figure 1 shows the state

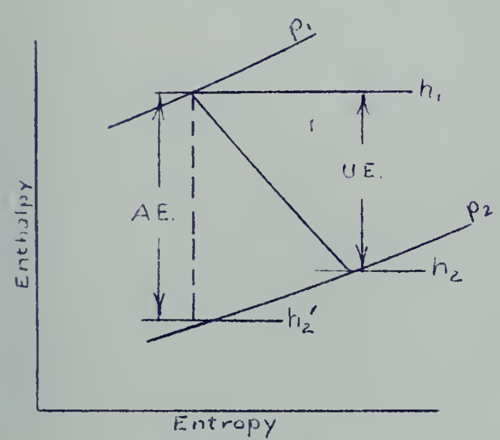

Figure 1

Unthrottled Turbine

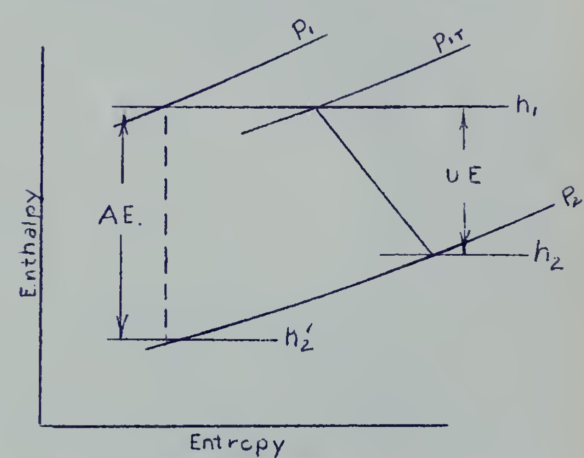

Figure 2

Throttled Turbine

line of an unthrottled turbine, and figure 2, a throttled turbine. Since the efficiency is the used energy divided by the available energy, (1)[1], $\left(\eta_T = \frac{U.E.}{A.E.}\right)$, it is obvious that the throttled turbine must use more steam than the unthrottled turbine to provide the same power output.

The initial step in the development of the theoretical machinery plant was the preliminary design of

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[1]Numbers in parentheses refer to references on page

a series turbine. Prerequisite to designing the series turbine, however, is the power required to drive the forced draft blowers. It was therefore necessary to design a centrifugal blower which would be suitable for use with the series turbine, coupled directly to the turbine shaft. The power required to drive the fuel oil service pumps is much less than that needed to drive the blowers (less than 2% at full power) and therefore a fuel pump was not designed, but an efficiency of 80% was assumed as a reasonable estimate, realizing that a very large error in pump efficiency would have a negligible effect on the power required from the series turbine.

Having a set of characteristics for the series turbine, it was possible to begin making heat balances. It was decided to computerize the heat balances, since a heat balance is an iterative process. The computer program (Appendix F) is designed for the G.E. time sharing console located at Webb Institute, and is written in FORTRAN IV.

A set of preliminary heat balances for 15,20,25,30 knots and full power were first computed, from the results of which the forced draft blower and the series turbine were redesigned. A final set of heat balances was then computed. The results of both preliminary and

final heat balances are included in this report.

Much of the data used in the original design heat balances (computed for the Navy by Gibbs and Cox, Inc.) were not usable after the installation of the series turbine. Although the original boilers generated steam at a pressure of 1200 psi at the superheater outlet at all steaming rates, the fixed nozzle area of the series turbine causes the drum pressure to vary over a comparatively wide range. Consequently the pressures at the superheater outlet and thus to the propulsion turbines are, at all steaming rates, much lower than in the design heat balances. It was necessary, therefore, to construct new state lines for the propulsion turbines.

It is general practice in the Navy, which uses many auxiliary turbine driven pumps and blowers, to keep the auxiliary exhaust pressure as low as practicable to minimize steam consumption. The auxiliary exhaust is utilized to heat the feed water in the deaerating feed tank, the pressure in which is kept at 15psig, fixing the maximum temperature at 250°F (without flashing the feed into steam), (2) . However, with the largest contributer of auxiliary exhaust (the forced draft blowers) being driven by the series turbine the quantity of

auxiliary exhaust steam to heat the steam is insufficient to meet the feed heating needs. The most efficient method to obtain steam is to extract steam at low pressures from the propulsion turbines. Hence a single bleed system is used for feed heating in the DFT. In addition, because the only major auxiliaries now being driven by auxiliary turbines are the main feed pumps and the lube oil service pumps, raising the auxiliary exhaust pressure to 31 psig is much less detrimental than it would be in the original DLG-26 plant; the increase in steam rate of the auxiliary turbines is more than offset by the increase in overall plant efficiency caused by increasing the maximum feed temperature to 276.8°F. The higher the feed temperature the less energy to be added by the fuel, and the heating of feed in a heater is at 100% efficiency compared with less than 90% efficiency of energy transfer in the boilers, (3). Thus the addition of the series turbine indirectly improves efficiency by permitting a higher feed temperature.

The changes in the plant parameters discussed in the preceding two paragraphs necessitated the calculation of many new temperatures, pressures, and enthalpies for the heat balances. The origins of all revisions are given in the Heat Balance sections.

PREPARATION OF PRELIMINARY HEAT BALANCES

FORCED DRAFT BLOWER

Type	Centrifugal plate fan, double suction
Shaft diameter	d_s = 8 in.
Eye diameter	d_o = 23.2 in.
Eye velocity	C_o = 100 fps
Flow	Q = 61,800 CFM
Vane inlet diameter	d_1 = 30 in.
Velocity at vane inlet	C_1 = 100 fps
Impeller inlet width	b = 8.29 in.
Vane inlet angle	β_1 = 13.23°
Number of vanes	Z = 24
Blade speed (inlet)	u_1 = 425 fps
Outside diameter of impeller	d_2 = 39 in.
Blade speed (outlet)	u_2 = 552 fps
Vane outlet angle	β_2 = 41.53°
RPM	N = 3250 RPM
Head	H = 4865 ft
Efficiency	η = 61.1%

The design point for the forced draft blower is the maximum boiler overload. For the preliminary design the draft loss was taken from the boiler technical manual, NAVSHIPS 351-0739, as 70.1 inches of water. The design was based on the required head and

volume flow rate to satisfy the draft loss and to support combustion. Appendix C contains the detailed calculations.

SERIES TURBINE

As in the case of the forced draft blower, the design point was maximum boiler overload. The design was based upon the power required to drive the blowers and fuel oil pump, and upon the weight flow of steam at maximum overload as listed in NAVSHIPS 351-0739. The turbine is a single stage, impulse turbine with two rows of moving blades, velocity compounded. It will provide 1100 horsepower at maximum overload with a drop in steam enthalpy of 14.1 $\frac{\text{Btu}}{\text{lb.}}$ (Appendix D).

Nozzle Height	L_N = 1 in.
Nozzle Chord	C_N = 1 in.
Blade height	L_B = 1 in.
Blade Chord	C_B = 1 in.
Mean blade diameter	d_m = 15 in.
Shaft diameter	d_s = 4 in.
Nozzle coefficient	K_n^2 = .893
Nozzle Angle	α = 12°
Nozzle Area	A_N = 3.339 in ^{2}
Number of Nozzles	N = 4

Nozzle width	$o = .835$ in.
Nozzle admission fraction	$\epsilon = .3785$
Nozzle Pitch	$p = 4.46$ in.
Circumferential nozzle width	$\tau = 4.02$ in.
First row exit area	$A_{B1} = 5.175$ in.^{2}
Fixed row exit area	$A_{Bf} = 8.64$ in.^{2}
Second row exit area	$A_{B2} = 15.02$ in.^{2}
First row entry angle	$\beta_1 = 15^\circ$
First row exit angle	$\gamma_1 = 18.8^\circ$
Fixed row entry angle	$\beta_2 = 26.88^\circ$
Fixed row exit angle	$\alpha_2 = 32.50$
Second row entry angle	$\beta_3 = 58.80$
Second row exit angle	$\gamma_2 = 69.5$
Efficiency	$\eta_t = 57.3\%$
Number of labyrinths	$n = 36$ (each end)
Gland leakoff, full power	$W_L = 3420$ lb/hr.
Gland leakoff, cruising	$W_L = 3550$ lb/hr.

PROPULSION TURBINE STATE LINES

The system pressures in the series turbine plant are controlled by the nozzle area of the series turbine. Even the boiler drum pressure is established by the series turbine nozzle area. The one property of the system which can be calculated from the conditions at any particular operating point is the specific volume

of the steam leaving the series turbine nozzles. Knowing the nozzle area, the velocity of the steam leaving the nozzles, and the amount of steam flowing, the specific volume can be calculated from the continuity equation,

$$v = \frac{A_N C}{W_S}$$

Because the steam in the boiler drum is always saturated, and because the velocity of steam leaving the series turbine nozzles is equivalent to calculable enthalpy change, the boiler drum pressure can be established, as long as the nozzle coefficient of the series turbine is known. A flow equation from which to calculate the isentropic enthalpy change was developed (Appendix B), but the change in enthalpy at high steaming rates in the saturated zone was too great to be accurately calculated by an equation. It was necessary therefore, to approach the problem graphically. The method utilized to establish the system pressures is as follows (see Appendix E for calculations):

1. The required brake horsepower output of the series turbine is found by determining the required fan horsepower and fuel pump horsepower, taking their sum, and applying an assumed turbine mechanical efficiency of 98%.

2. The available energy required for the turbine is found, assuming that the efficiency established for the boiler maximum overload condition is constant for all steaming conditions.

3. The nozzle exit velocity, C_1, is calculated by finding the enthalpy drop across the nozzles based on the nozzle coefficient found in the series turbine design.

4. Using an approximate flow of steam, the specific volume at the turbine nozzle exit is determined from the continuity equation.

5. From the Ellenwood and Mackey thermodynamic charts a plot is prepared showing the enthalpy drop from the saturation line to various specific volumes, along lines the slope of which is fixed by the nozzle coefficient K_N^2, for various drum pressures (Figure 3).

6. Figure 3 is entered with specific volume and nozzle enthalpy drop to determine the boiler drum pressure.

7. A series of parallel lines, defined by the nozzle coefficient, are drawn on the Ellenwood and Mackey charts. Knowing the drum pressure and the enthalpy drop across the nozzle the series turbine exit pressure can be found.

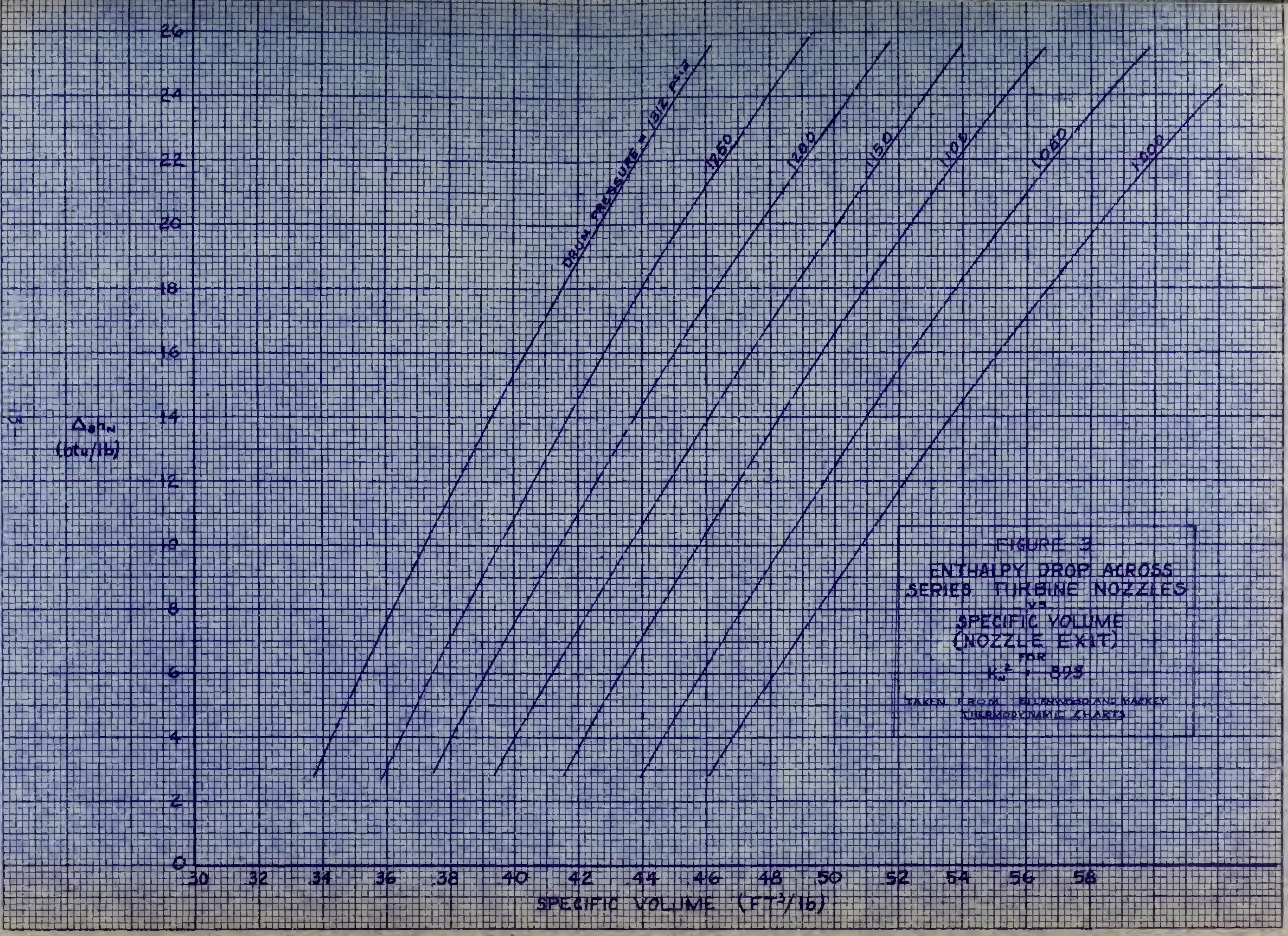


FIGURE 3

ENTHALPY DROP ACROSS SERIES TURBINE NOZZLES vs SPECIFIC VOLUME (NOZZLE EXIT) FOR $K_N^2 = .893$

TAKEN FROM KEENAN AND KEYES THERMODYNAMIC CHARTS

8. The pressure drop through the superheater is calculated from the pressure drop of the original design. The pressure drop is proportional to the square of the steam flow and the specific volume; that is

$\Delta p_1 = KW_1^2 v_1$ for the original design,

and $\Delta p_2 = KW_2^2 v_2$ for the series turbine design.

Then,

$$\frac{\Delta p_2}{\Delta p_1} = \frac{KW_2^2 v_2}{KW_1^2 v_1},$$

or

$$\Delta p_2 = \left(\frac{W_2}{W_1}\right)^2 \left(\frac{v_2}{v_1}\right) \Delta p_1$$

9. The pressure drop in the line is found in the same manner as that through the superheater. The pressures in the line are considerably lower with the series turbine than in the original design, and consequently the specific volumes are higher, causing a greater line pressure drop than could be tolerated. Therefore it was necessary to increase the main steam piping diameter from six to eight inches. Consequently the line drop is given by

$$\Delta p_2 = \left(\frac{W_2}{W_1}\right)^2 \left(\frac{v_2}{v_1}\right) \left(\frac{D_1}{D_2}\right)^4 \Delta p_1 .$$

10. The first stage inlet pressure is found by multiplying the pressure drop across the throttle in the original design heat balances by the ratio of the

throttle pressure over the throttle pressure in the original design.

The propulsion turbine state lines of the original ship were drawn from the temperature and pressure data presented in the Propulsion Turbine Technical Manual, NAVSHIPS 341-1346. Using the first stage inlet pressures found as in 10 above, and initial enthalpies as published in the original design heat balances, the revised state lines were drawn parallel to the original state lines.

Because the propulsion turbine inlet pressures are lower with the series turbine than in the original ship, modifications must be made to the propulsion turbines. It is possible, however, to modify only the first stage to accomodate the higher specific volume of steam. If the first stage exit pressure is maintained about the same as in the original ship then no changes to the succeeding stages are necessary. The nozzles and blades of the first stage must be enlarged to pass approximately the same steam flow at a higher specific volume, and to reduce the enthalpy drop across the first stage so as to achieve the desired first stage exit pressure. A relationship for determining the approximate exit pressure can be established by noting that

$W_s = p_1 QRA_N.$

Now, (4)

$$Q = f(1/\sqrt{T}) = K/\sqrt{h_1^{\circ} - C}$$

and

$$R = f(p_2/p_1) = \frac{r\sqrt{K_N^2(1 - r^{\frac{k-1}{k}})}}{1 - K_N^2(1 - r^{\frac{k-1}{k}})}.$$

Therefore,

$$W_s = p_1\left(\frac{1}{\sqrt{T}}\right)\left(\frac{p_2}{p_1}\right)A \quad \text{for the original ship}$$

and $W_s' = p_1'\left(\frac{1}{\sqrt{T'}}\right)\left(\frac{p_2'}{p_1'}\right)A'$ for the modified turbine,

then $\frac{W_s A'}{W_s' A} = \sqrt{\frac{T'}{T}}\left(\frac{p_2}{p_2'}\right).$

As a close approximation let

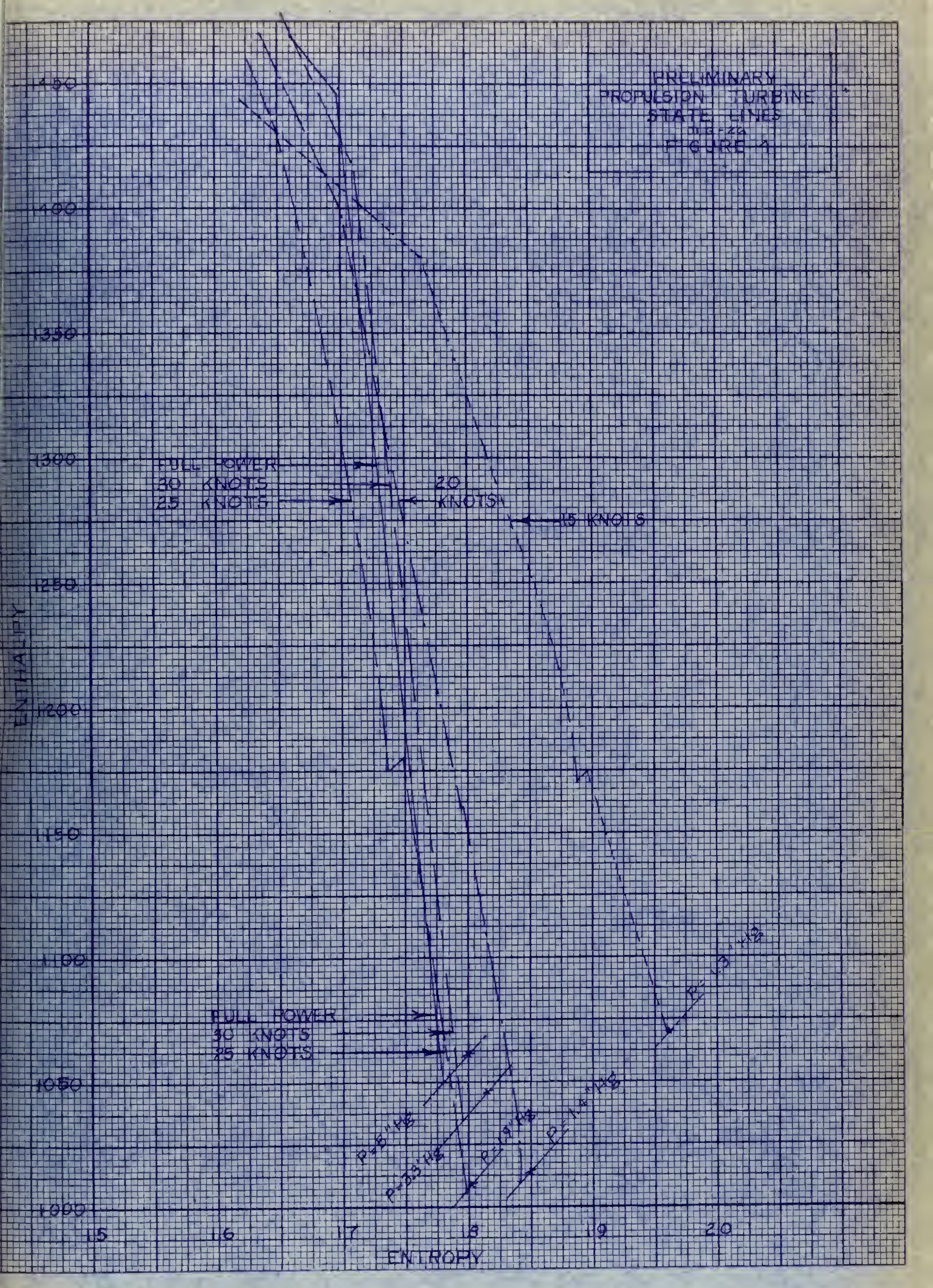
$$\frac{W_s A'}{W_s' A} = 1;$$

then $p_2' = p_2\sqrt{\frac{T'}{T}}.$

The leaving losses for the propulsion turbines are not published, but knowing the state line end points, and assuming that the flow leaving the sixth (final) stage of the l.p. turbine is always axial, an approximate leaving loss can be calculated, by first finding the blade exit area from

$A = \pi d_m L \epsilon$, where L = 9 in.

Then the exit velocity can be found from the continuity equation.


PRELIMINARY
PROPULSION TURBINE
STATE LINES
FIGURE 4
1450
1400
1350
1300
1250
1200
1150
1100
1050
1000
ENTHALPY
1.5
1.6
1.7
1.8
1.9
2.0
ENTROPY
FULL POWER
30 KNOTS
25 KNOTS
20 KNOTS
15 KNOTS

The preliminary state lines are shown in figure 4 and the turbine exit condition is indicated by a small "x" on the line of constant pressure corresponding to the condenser pressure at the particular operating condition.

HEAT BALANCES

The heat balance computer program and the input data for the computer program are contained in the confidential Appendices F and G. However, the methods of obtaining the input data are described here.

1. The initial flow from the main condenser was calculated from the original flow using the relationship

$$Q = Q_o \frac{(U.E.)_o}{(U.E.)}$$

2. The mechanical efficiency is not published, but it can be calculated for each steaming rate from the equation

$$\eta_m = \frac{2544 \times SHP}{W_s \times U.E.}$$

3. The superheater and desuperheater outlet enthalpies are taken as the same as in the original heat balances, and the temperatures are as fixed by the pressures and enthalpies.

4. The enthalpies at the propulsion turbine inlet were taken as those leaving the superheater, retaining the practice which was used in the original heat balances, in which the line enthalpy losses were neglected.

5. The flow from the auxiliary condenser was assumed to be the same as in the original design heat balances for the case of the preliminary heat balance only.

6. The temperatures leaving the condensers were taken as the same as those for the original heat balances.

7. The enthalpy of the propulsion turbine gland leakoff was calculated for each steaming condition from data given in NAVSHIPS 341-1346. It was found that in the original heat balances the quantities of leakoff steam were increased by 30%, apparently to account for wear, (5). To maintain consistency between these and the original heat balances the same 30% increase was used in the heat balances with the series turbine.

8. The gland leakoff enthalpy of the series turbine was calculated in the section on series turbine design. The quantity of steam was found from the expression, (6)

$$W_L = A\alpha\phi\gamma\sqrt{g\frac{p_1}{v_1}} \; .$$

9. In order to provide sufficient steam for feed heating, steam will be bled from one of three points in the propulsion turbine, depending upon the ship's speed. If the speed is increased from "stop" to full power, bled steam is first required at about thirteen knots, and it is first bled from the sixth stage outlet of the hp turbine. With continued acceleration the bleed point becomes the 8th stage outlet when the pressure there becomes 48 psia. As speed continues to increase the crossover pressure increases until it reaches 50 psia, at which time it becomes the bleed point.

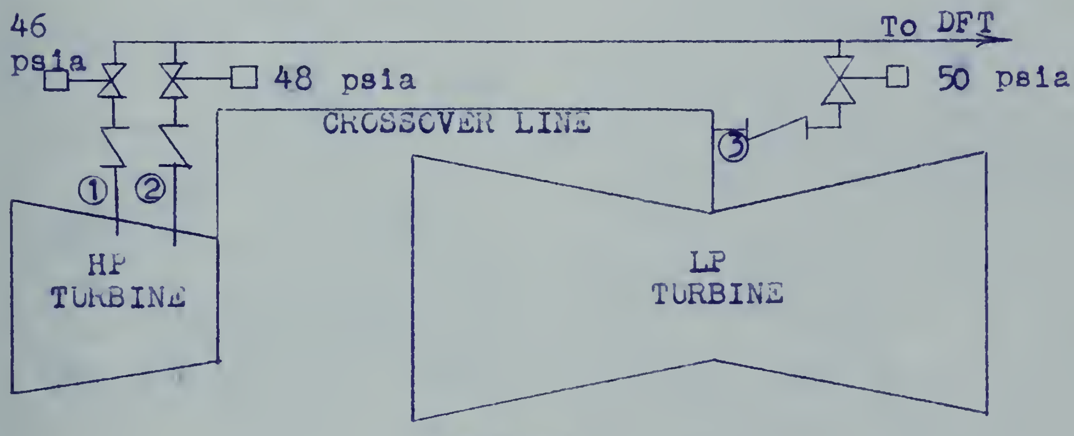


Figure 5

Propulsion turbine bleed system

Bleed point 1 : 6^{th} stage exhaust
56.2 psia at 15 knots. h = 1310.6 btu/lb.
Bleed point 2 : 8^{th} stage exhaust
60.7 psia at 20 knots. h = 1269.4 btu/lb.
124.0 psia at 25 knots. h = 1294.0 btu/lb.
Bleed point 3 : Crossover line
51.1 psia at 30 knots. h = 1246.2. btu/lb.
73.4 psia at full power. h = 1268.7 btu/lb.

Owing to the extreme difficulty of programming the turbine state lines, the bleed pressure is calculated by the computer and printed, and the bleed enthalpy must be taken from the state line and inserted into the computer for each iteration.

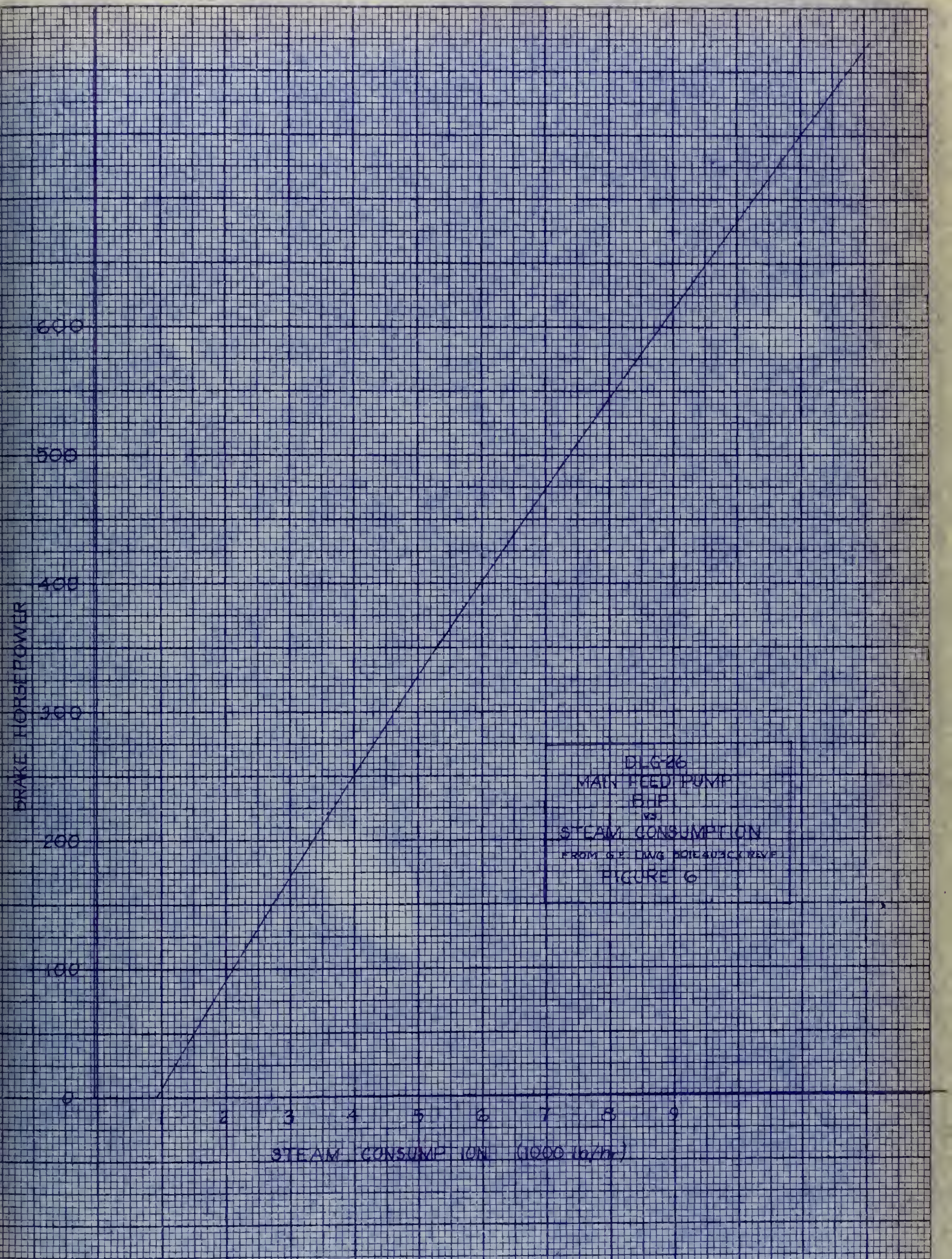
10. The main feed pump steam consumption is dependent upon the feed flow, and so varies slightly with each iteration. In order to utilize the computer it was necessary to plot a curve of steam consumption vs. feed flow, and, by regression analysis, fit an equation to the curve which could be used in the program. It was necessary to plot first the pump BHP vs. steam consumption (Figure 6), and then convert BHP to feed flow (Figure 7). The equation obtained is

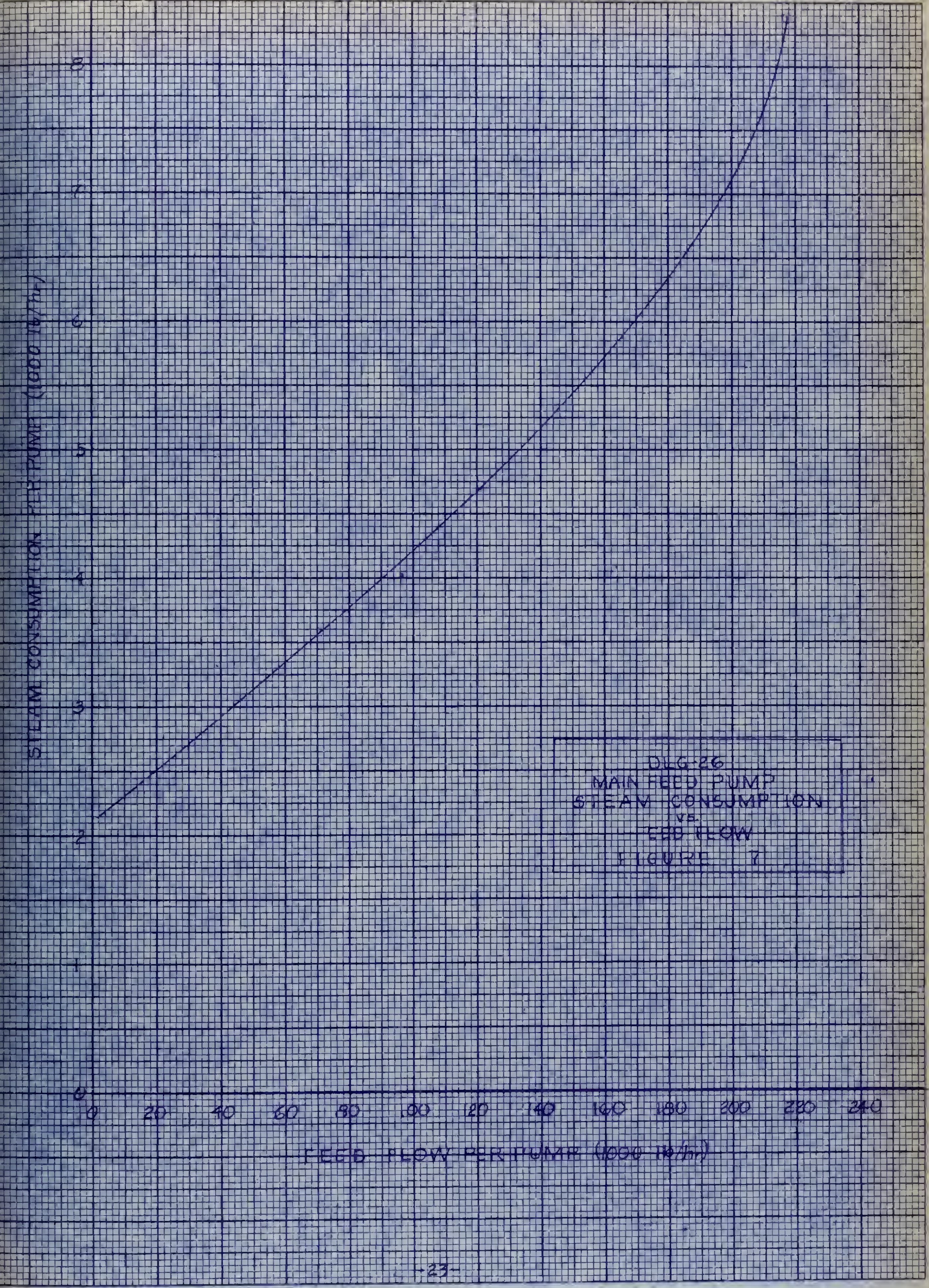
$$y = (2.655)(10^{-8})(x^2) + (0.01839)(x) + 2120.$$

The error in using the equation is less than 1% (.684%) at about 30 knots.

Because the exhaust pressure for the auxiliaries is higher than in the original heat balances, the steam consumption for the main feed pump had to be corrected by a factor,

$$\frac{\text{A.E. original}}{\text{A.E. Revised}}.$$


BRAKE HORSEPOWER
0
100
200
300
400
500
600
2
3
4
5
6
7
8
9
STEAM CONSUMPTION (1000 lb/hr)
DLG-26
MAIN FEED PUMP
BHP
vs
STEAM CONSUMPTION
FIGURE 6



DLG-26
MAIN FEED PUMP
STEAM CONSUMPTION
vs.
FEED FLOW
FIGURE 7
STEAM CONSUMPTION PER PUMP (1000 lb/hr)
FEED FLOW PER PUMP (1000 lb/hr)
0
1
2
3
4
5
6
7
8
0
20
40
60
80
100
120
140
160
180
200
220
240

Since the boiler drum pressure is lower with the series turbine than in the original ship the feed pump output pressure need not be as high. Therefore the steam consumption had to be further corrected by a factor,

$$\frac{p_{S.T.} + 25}{p_c + 25},$$

where

$p_{S.T.}$ = drum pressure with series turbine,

p_o = drum pressure in original ship,

and 25 = pressure of MFP outlet over that of drum.

The enthalpy of the main feed pump exhaust was calculated using the expression

$$h_{out} = h_{in} - \Delta h = h_{in} - \frac{BHP\ (2544)}{W_s}.$$

11. In ascertaining the fuel oil heating steam flow it was assumed that the oil is heated from 75°F to 150°F. The equation,

$$Q = \frac{34.45\ W_f}{h_{STLO} - 218}$$

was made a part of the computer program so that an exact amount of steam could be calculated in each iteration. The expression is reduced from, (7)

$$Q = \frac{W_f C_p T}{h_g - h_f} = \frac{W_f(0.46)(75)}{h_{STLO} - 218} = \frac{34.45\ W_f}{h_{STLO} - 218},$$

where h_g = Enthalpy of steam entering,

h_f = Enthalpy of saturated water at 250°F,

and h_{STLO} = Enthalpy of series turbine leakoff.

12. Drain temperatures are in accordance with S.N.A.M.E. Bulletin T & R 3-11, except that the main feed pump gland cooling water is assumed to increase in enthalpy 10 btu/lb across the glands, (8). The enthalpy of the main feed pump gland leakoff is 200 btu/lb less than the throttle enthalpy, (9). The main air ejector inter condenser drain temperature cannot be at 125°F as specified in T & R 3-11, but it will vary depending upon the steaming rate. It will be established by adding to the enthalpy of the condensate from the condenser the same enthalpy difference that is prescribed in T & R 3-11 for 1½ in Hg.

13. The drains from the fuel oil heater in the original heat balances are piped to the deaerating feed tank. In order to avoid the danger of introducing fuel into the feed water the drains from the fuel oil heaters have been rerouted to the low pressure drain tank.

14. Operating experience in both the Navy and the Merchant Marine has shown that steam atomizing burners are far superior to mechanical atomizing burners. Whereas in the original ship fuel oil pressures of 1148 psig were required for proper atomization with the pneumatic combustion control system, a change to steam atomization permits fuel pressures of 350 psi in the series turbine ship. The steam required for atomization is calculated in the computer program from the

relationship,

$$Q = \frac{\text{SHP x Fuel rate}}{20}$$

15. The gland leakoff from the series turbine is utilized in the main and auxiliary air ejectors, propulsion turbine gland seal, evaporator air ejectors, fuel oil heating, and steam atomization; and any excess is reduced to 31 psig and routed to the deaerating feed tank.

16. Steam for ship heating and for galley, laundry, and hot water is taken from the deaerating feed tank.

17. The auxiliary gland leakoff condenser in the original ship uses sea water as a coolant. In the interest of efficiency condensate is used as a coolant in the series turbine ship.

The preliminary heat balances are calculated by computer. In order to prepare the state lines and input data for the final heat balances only the feed to the boiler and the condensate leaving the condensers are needed from the preliminary heat balances; and for that reason the preliminary heat balances are not presented on a flow diagram, but rather the results are tabulated in Appendix H.

PREPARATION OF FINAL HEAT BALANCES

FORCED DRAFT BLOWER

Following the preparation of the preliminary heat balances it was discovered that the published draft losses in the Main Boiler Technical Manual, NAVSHIPS 351-0739, did not include the draft loss through the uptakes. Consequently it was necessary to redesign the forced draft blower to provide it with the ability to develop the head necessary to overcome the total draft loss. Appendix A contains the detailed calculations, and a summary of the final forced draft blower characteristics follows:

Shaft diameter	d_s = 8 in.
Eye diameter	d_o = 23.2 in.
Vane inlet diameter	d_1 = 30 in.
Impeller inlet width	b = 8.29 in.
Vane inlet angle	$\beta_1 = 12.5^\circ$
Number of vanes	Z = 24
Outside diameter of impeller	d_2 = 39 in.
Vane outlet angle	$\beta_2 = 42.1^\circ$
RPM	n = 3450 rpm
Efficiency	η = 62.1 %

Fan type: Centrifugal plate fan
double suction.

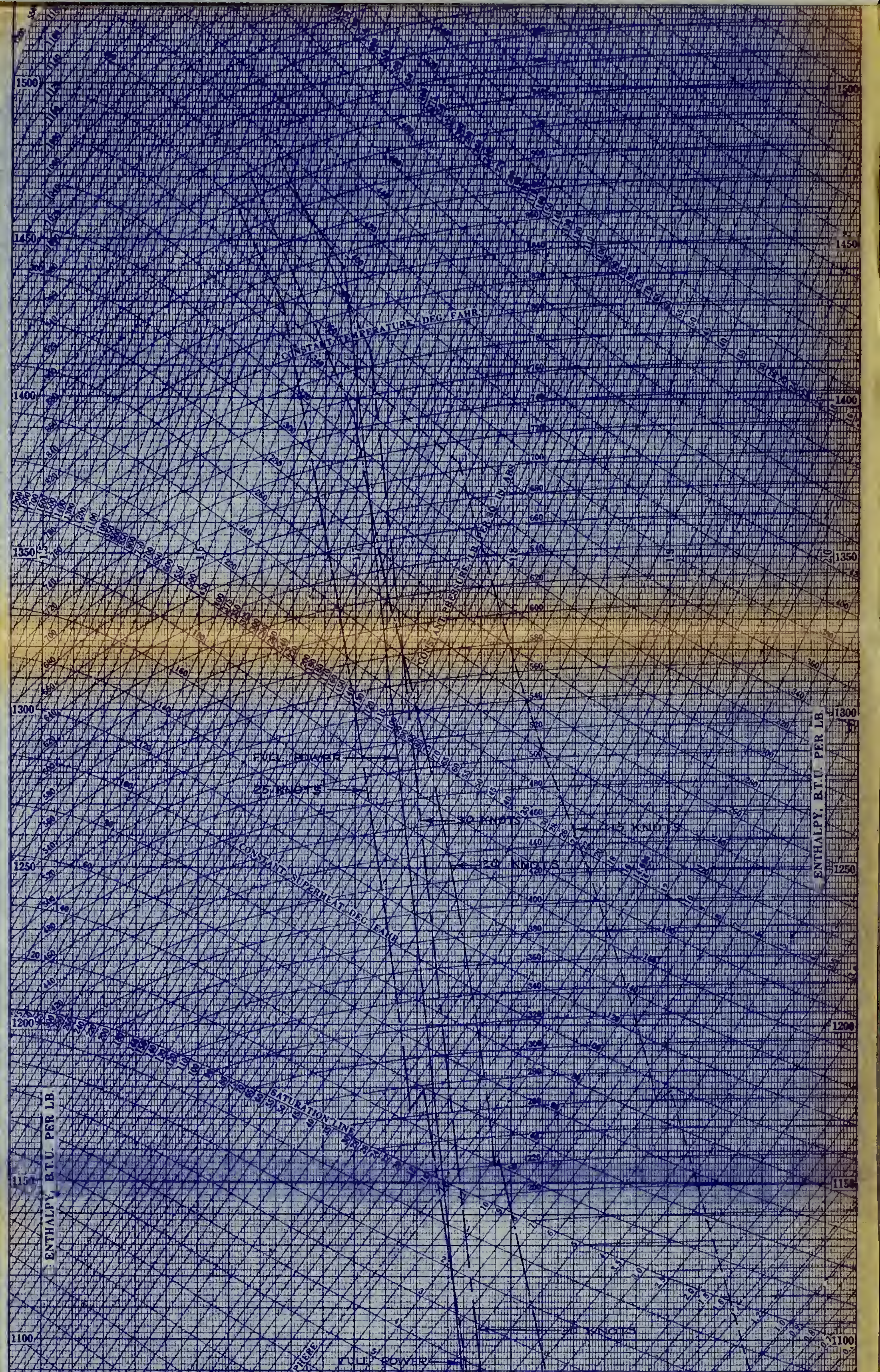
SERIES TURBINE

The increased head of the forced draft blower imposes a greater power demand upon the series turbine, requiring it to be redesigned. It became necessary for the series turbine to produce 1225 hp at maximum (120 %) boiler overload as compared with the 1100 hp of the preliminary design. The calculations are contained in Appendix B.

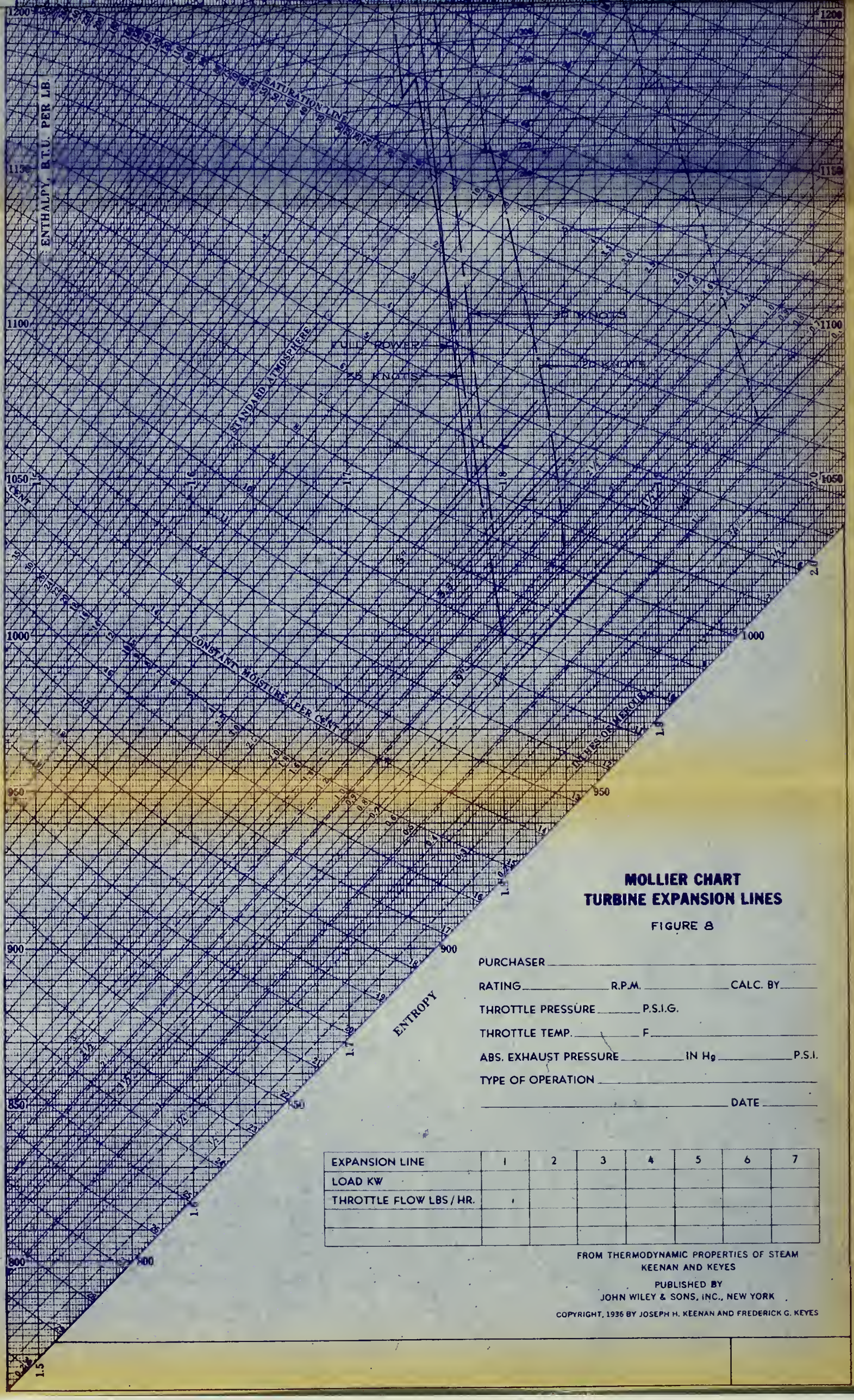
Nozzle and blade height	$L = 1$ in.
Nozzle and blade chord	$c = 1$ in.
Mean blade diameter	$d_m = 15$ in.
Shaft diameter	$d_s = 6\ 3/4$ in.
Nozzle coefficient	$K_N^2 = .89$
Nozzle angle	$\alpha_1 = 12^\circ$
Nozzle area	$A_N = 3.21$ in.^{2}
Number of nozzles	$N = 4$
Nozzle admission fraction	$\epsilon = .364$
First row exit area	$A_{B1} = 4.98$ in.^{2}
Second row exit area	$A_{B2} = 14.48$ in.^{2}
First row entry angle	$\beta_1 = 14.92^\circ$
First row exit angle	$\gamma_1 = 18.82^\circ$
Fixed row entry angle	$\beta_2 = 26.77^\circ$
Fixed row exit angle	$\alpha_2 = 32.3^\circ$
Second row entry angle	$\beta_3 = 57.9^\circ$
Second row exit angle	$\gamma_2 = 69.2^\circ$
Efficiency	$\eta_T = 56.6$ %
Number of labyrinths	$N = 36$ (Each end)

Propulsion Turbine State Lines

The revision in the nozzle area of the series turbine causes the system pressures to change from those determined for the preliminary state lines. Because the nozzle area is smaller than in the preliminary design, it would be expected that the system pressures would be lower. However, the final design was based upon the steam flow found in the preliminary design, which is significantly lower than that of the original ship. The result of the combination of decreased area and decreased steam generation is higher system pressures throughout. The calculations for the final state lines are found in Appendix E, and the state lines are presented in Figure 8.



1500
1450
1400
1350
1300
1250
1200
1150
1100
CONSTANT TEMPERATURE DEG. FAHR.
CONSTANT PRESSURE LB. PER SQ. IN. ABS.
FULL POWER
25 KNOTS
30 KNOTS
15 KNOTS
20 KNOTS
CONSTANT SUPERHEAT DEG. FAHR.
SATURATION LINE
ENTHALPY, B.T.U. PER LB.
FULL POWER



MOLLIER CHART
TURBINE EXPANSION LINES

FIGURE 8

PURCHASER ____________________

RATING __________ R.P.M. __________ CALC. BY __________

THROTTLE PRESSURE ______ P.S.I.G.

THROTTLE TEMP. ______ F ______________

ABS. EXHAUST PRESSURE ________ IN Hg ________ P.S.I.

TYPE OF OPERATION ____________________

______________________________ DATE __________

EXPANSION LINE	1	2	3	4	5	6	7
LOAD KW							
THROTTLE FLOW LBS / HR.							

FROM THERMODYNAMIC PROPERTIES OF STEAM
KEENAN AND KEYES

PUBLISHED BY
JOHN WILEY & SONS, INC., NEW YORK

HEAT BALANCES

The final heat balances were prepared in the same manner as were the preliminary heat balances, the computer input data being determined on the basis of the preliminary heat balances. The heat balances are contained in Appendix I.

The table below presents a comparison between the fuel rates for the original ship and those for the ship modified with the series turbine.

	Full Power	30 Knots	25 Knots	20 Knots	15 Knots
Original ship	.611	.583	.588	.667	1.004
Series turbine ship	.589	.563	.587	.648	.944
Difference	.022	.020	.001	.019	.06
Percent Improvement over Original Ship	3.6	3.43	.17	2.84	5.97

The flow of condensate leaving the main condenser in the final heat balance is sufficiently close to that found in the preliminary heat balances that the system pressures need not be reevaluated for a third set of turbine state lines. The final heat balances themselves are each the result of six iterations per steaming condition.

PROPOSED MACHINERY ARRANGEMENT

It is of little value to propose a marine engineering innovation if the resulting machinery arrangement is impractical for the ship for which it is intended. In the case of the series turbine the alterations to the existing machinery plant involve only the forced draft blowers, the fuel oil pumps, and the boilers.

The existing forced draft blowers must be replaced with those designed for the series turbine. The blower shafts are coupled directly to the series turbine shaft, one on either end of the series turbine, and the turbine and the blowers are located as near to the boiler drum as possible. For DLG-26 it is proposed that the location be as is shown in Figures 9 and 10. As shown, the unit (series turbine and blowers) is supported by frames built up from I-beams, the bases of which are welded to the existing 6 5/8 inch downcomers. The proposed arrangement permits easy access for maintenance from the upper level of the fireroom where there is a walkway between the side of the boiler and the feed pumps. The unit is high enough above the boiler casing that the existing inspection ports and access panels above the boiler furnace are unobstructed. For the proposed arrangement

the air ducting must be modified to provide air to both of the forced draft blowers, as can be seen in Figure 10.

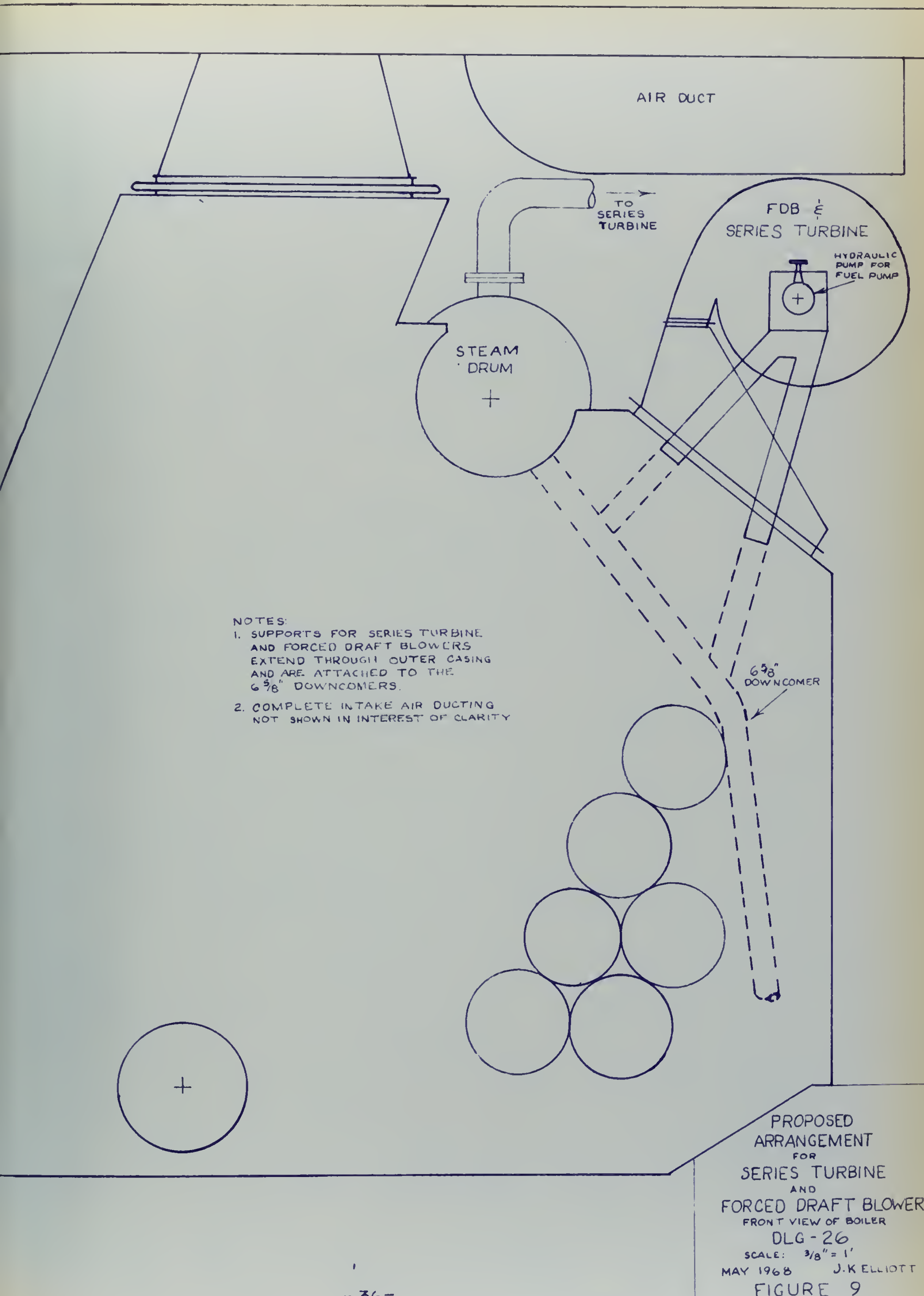
In the original ship the fuel oil service pumps are located on the lower level of the firerooms with their suctions very low in the ship. The pumps are screw type positive displacement pumps and they do not draw sufficient vacuum for them to be moved to a location higher in the ship. To drive the fuel oil service pumps from the series turbine it is necessary to transmit the energy of the turbine to the pumps on the lower level. The proposed arrangement is to drive the pumps with hydraulic motors, the pumps for which are coupled to the shafts of the forced draft blowers as is shown in Figure 10. The hydraulic pumps are to be fitted with pneumatic remote operating tilt box positioners for the purpose of adjusting the fuel-air mixture as is required to compensate for the varying qualities of different batches of fuel.

The fuel oil delivery to the burners in the original ship is regulated by a pneumatically actuated combustion control system. Oil is delivered at a pressure of 1147.5, (10), to the burners where the control system admits a portion of the oil to the

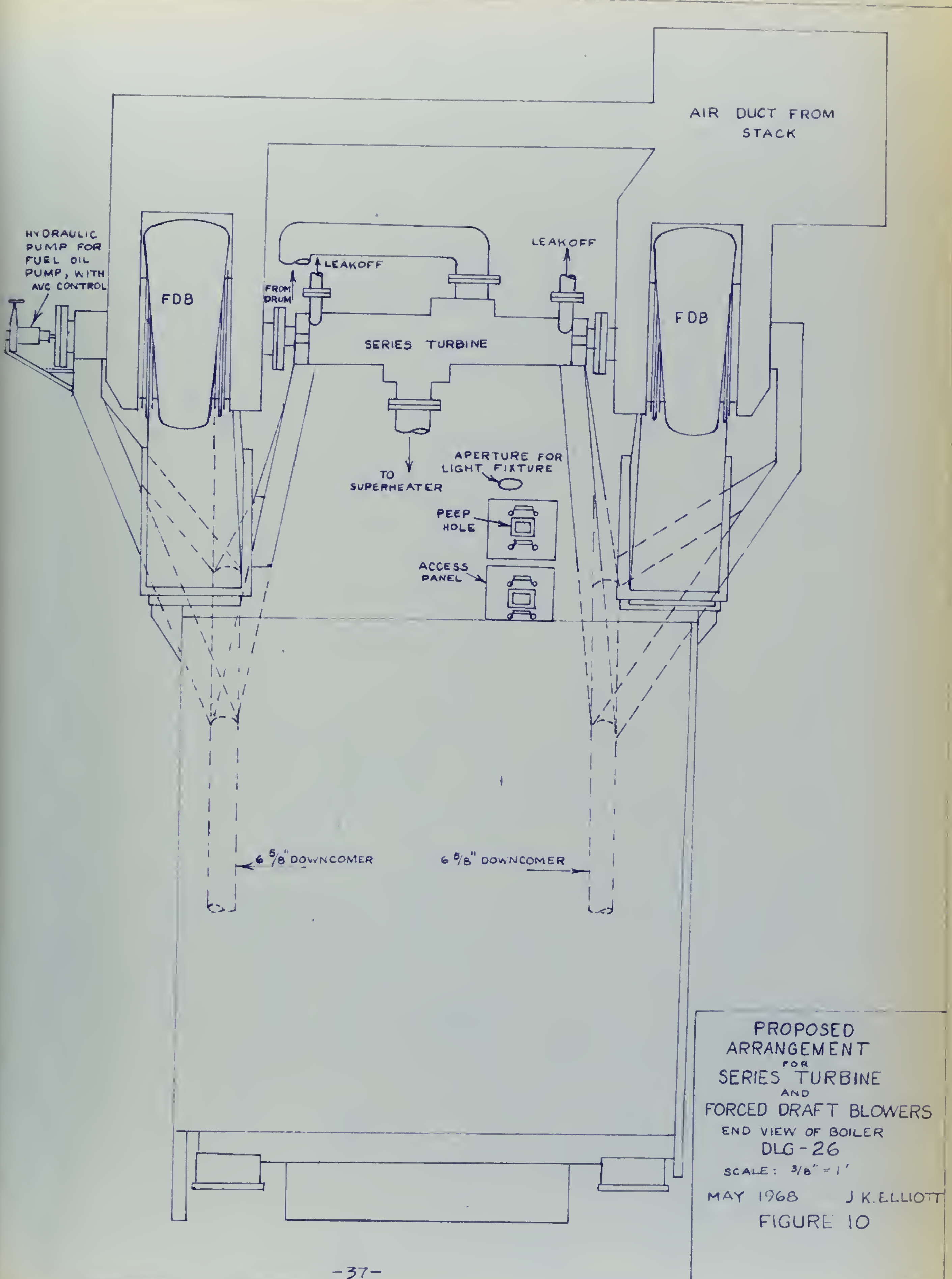
boiler and recirculates the remainder of the oil back to the pump inlet. Fuel Atomization is done mechanically, and in order to achieve a satisfactory oil spray the high pressure (1147.5 psig) is required. With the series turbine system, however, steam atomization has been assumed, where the atomizing steam is supplied by the series turbine gland leakoff. Steam atomization furnishes an excellent spray at much lower pressures than is used in the original ship. Selecting a pressure of 350 psig the required brake horsepower to drive each fuel oil service pump is only 8.5 hp at maximum boiler overload, far less than the 73.5 rated BHP of the fuel oil service pumps in the original ship,(11). In adapting the DLG-26 to the series turbine, then, the fuel oil pumps should be replaced with pumps more suitably designed for the revised requirements.

The lighting off forced draft blower is located in the original ship where the series turbine is positioned in the proposed arrangement. Therefore it should be relocated to the area on the economizer side of the boiler where, in the original ship, a main forced draft blower is installed.

An electrical l.p. steam generator must be installed to provide approximately 100 pound per hour of steam at 150 psig for lighting off purposes when starting from a completely dead plant. The electrical power required will be 25 kw.



PROPOSED ARRANGEMENT FOR SERIES TURBINE AND FORCED DRAFT BLOWER
FRONT VIEW OF BOILER
DLG-26
SCALE: 3/8" = 1'
MAY 1968 J.K ELLIOTT
FIGURE 9



PROPOSED
ARRANGEMENT
FOR
SERIES TURBINE
AND
FORCED DRAFT BLOWERS
END VIEW OF BOILER
DLG-26
SCALE: 3/8" = 1'

MAY 1968 J.K. ELLIOTT

FIGURE 10

AUTOMATIC COMBUSTION CONTROL

The purpose of this section is to describe the processes by which the series turbine system automatically adjusts the amount of fuel oil and air supplies to the boiler to generate the required amount of steam. In general terms, whenever the turbine steam demand is greater than is being provided by the boiler, the series turbine increases speed, thus supplying more air and fuel, until the demand of the propulsion turbine is met, at which time the series turbine speed becomes steady at the new steaming level.

The changes in speed of the series turbine are due to variations in the flow rate of the steam. For example, if it is desired that the ship's speed increase, the throttleman opens the throttle to admit an increased steam flow to the turbine. (In case of a sudden, large change of speed the line pressure may drop, creating an increased specific volume.) Immediately the exit velocity of the steam from the series turbine nozzles increases in accordance with the continuity equation,

$$C_1 = \frac{W_s\ v}{A_N}.$$

As can be seen, the nozzle exit velocity increases

due to increases in both weight rate of flow and specific volume. The increased nozzle exit velocity, C_1, creates an increased enthalpy change across the nozzles, defined by the expression,

$$C_1^2/2gJ = (\Delta h)_N.$$

The isentropic enthalpy change is increased, since

$$(\Delta h)_N = \eta_N \Delta_1 h;$$

and therefore the work of the turbine is increased:

$$wk = \eta_T \Delta_1 h = \frac{\eta_T (\Delta h)_N}{\eta_N} = \left(\frac{\eta_T}{\eta_N}\right)\left(\frac{C_1^2}{2gJ}\right).$$

The power output is similarly raised as is defined by

$$P = wk \times W_s.$$

Now the series turbine is an essentially constant specific volume device, and after a change in throttle setting a change in specific volume is only a transient effect which serves to accelerate the return to equilibrium at which point the specific volume will have returned to its approximate original value. With small speed changes the specific volume will not be effected, but in large abrupt changes line pressure variations may occur and specific volume fluctuations will result. But attributing the total change in power output to the change in the weight rate of flow, then the power increases as the third power of the weight rate of flow, W_s. That is,

$$P = wk \times W_s$$
$$= \left[\left(\frac{\eta_T}{\eta_N}\right)\frac{C_1^2}{2gJ}\right]W_s$$
$$= \left[\left(\frac{\eta_T}{\eta_N}\right)\frac{1}{2gJ}\left(\frac{W_s v}{AN}\right)^2\right]W_s$$
$$= K_1 W_s^3,$$

where K_1 = a constant.

Now the increased power delivered to the forced draft blower and fuel oil pump produces increased air and fuel flows in proportion to the cube root of the power. That is,

$$P = \frac{H \times W}{\eta},$$

or

$$W = \frac{P \times \eta}{H},$$

where, (12),

$$H \propto W^2,$$

so that

$$W^3 = K_2 P, \qquad K_2 = \text{a constant},$$

or

$$W = \sqrt[3]{K_2 P}.$$

Thus an increase in steam flow produces a proportional increase in fuel and air flows:

$$\Delta W_A \propto \Delta W_f \propto \Delta W_s.$$

Changes in air and fuel supplies which are proportional to the change in steam flow rate are precisely the proportions required by the boiler in

order to generate the desired steam flow. The weight rate of fuel is determined from the expression,

$$W_f = \frac{W_s(\Delta h)}{HHV \times \eta_B},$$

where

Δh = heat added per pound of steam in the boiler,

and

HHV = higher heating value of the fuel.

Thus,

$$W_f \propto W_s,$$

and

$$W_A = R \times W_f,$$

where

R = lb air/lb fuel.

Therefore,

$$W_A \propto W_f \propto W_s.$$

Thus it has been shown that an increased demand for steam creates a demand for a proportional increase in fuel oil and air. The series turbine, with an increased steam flow passing through it, generates a sufficient amount of power for the fuel pump and blowers to deliver the necessary quantities of fuel and air to sustain the new steaming rate. Thus the series turbine system provides complete automatic combustion control.

MANNING AND OPERATION

A reliable combustion control system as is afforded by the series turbine provides the capability of reducing the number of watch standers that are stationed in the firerooms. The watch could be stationed in a control booth in which are located the following devices:

Boiler drum level indicators

Boiler drum pressure and temperature gages

Superheater outlet pressure and temperature gages

Desuperheater outlet pressure and temperature gages

Outlet pressure gages for main feed pumps, booster pumps, fuel oil service pumps, and forced draft blowers.

Feed water temperature gage

Alarms for low and high drum water levels and DFT water level

Alarms for low lube oil pressure to pumps, blowers, and series turbines.

Remote control device for fuel oil pump hydraulic pump tilt box.

Actuating devicesfor main boiler stop air motor, and feed-check valve air closing motor.

The watch could consist of a boiler top watch and a messenger, who would be detailed to the upper level in event of a casualty.

In lighting off the boilers the air and fuel are initially supplied by the lighting off blower and the port use fuel oil service pump, and steam for atomization is supplied from the in port steaming boiler. In case of dead plant where none of the boilers are in operation, steam must be supplied by the small electrical steam generator (Requiring 25kw for 100 lb/hr steam at 165 psia). A line must be supplied from the steam drum to the superheater, bypassing the series turbine, to provide a flow through the superheater for protection of the tubes. In case of dead plant the drum water level must be maintained by using the electric reserve feed transfer pump, until sufficient steam is generated to drive a main feed pump. When the drum pressure increases to where the series turbine is being driven by its own gland leakoff steam the lighting off blower and port use fuel oil pump can be secured. Upon switching from the reserve feed transfer pump to a main feed pump the watch standers may take stations in the control booth and the lighting off procedure is completed.

In the event of a casualty to the boiler initial action can be taken by the top watch in the control booth. The actions that can be taken for specific casualties are as follows:

Loss of fuel oil suction.

The fires go out and the blowers continue to run, purging the furnace, until the drum pressure drops to where the series turbine stops turning or until fuel oil suction can be shifted to a full tank. The feed is regulated by the three element control system installed in the original ship.

Failure of fuel oil service pump.

The fires die out, as in loss of fuel oil suction. In this case, however, the boiler cannot be relighted. Therefore the plant must be crossconnected until the standby boiler can be lighted off and put on the line. The main boiler stop must be closed prior to cross-connecting the plant.

Boiler Tube or other Pressure Part Carries Away.

Adjust the fuel oil hydraulic pump tilt box to reduce the pressure to zero. With no fuel oil pressure the fires go out and the blowers continue to blow steam from boiler until series turbine stops running. The boiler stops must be closed in order to cross connect

the plant, at which time the lighting off blower should be started to continue to blow steam from the furnace.

Low Water.

Secure the fuel oil pressure to the boiler by adjusting the tilt box of the hydraulic pump. Close the main boiler stop and the feed-check valve by the remote actuated air motors. The plant must be cross-connected until a standby boiler can be brought onto the line.

High Water.

After the engine room is notified to close the guarding valve the feed-check valve must be closed by the remote actuated air motor, the fuel oil pressure reduced to zero by adjusting the hydraulic pump tilt box, and the main stop valve closed by the air motor. The drum water level is lowered by bottom blow or surface blow.

Fuel oil leak.

For a major fuel oil leak the fuel oil pressure drops and the fires go out. The boiler must be secured and the plant cross connected until a standby boiler is brought onto the line or until the leak is repaired.

Boiler Casing fire.

Close the main boiler stop valve and simultaneously stop fuel oil to the boiler. The boiler casing steam smothering system must be actuated to extinguish the fire.

As has been described above, any initial action required in event of a fireroom casualty can be taken by the watch stander in the control booth.

SUMMARY

Four areas have been considered in this study, each resulting in a favorable outcome. The heat balances have demonstrated that the series turbine plant is more efficient than the original plant at all steaming conditions. A reduction in fuel rate of 2.84% at 20 knots provides an increase in endurance of 250 miles, or it permits carrying 45 less tons of fuel oil, either of which is a desirable result. It has been shown that the revisions in machinery and arrangement can readily be accomodated by the existing ship and basic machinery arrangement, that only the lighting off forced draft blower need be relocated, and that a small l.p. steam generator is the only additional auxiliary equipment required. The system has been theoretically proven to be inherently automatic, a finding supported by the yet undocumented but partially successful experimental Maritime Administration series turbine boiler under test at the Naval Boiler and Turbine Laboratory. Reliable automation permits reduced manning for both operation and casualty control. The pneumatically actuated combustion control system presently installed in the DLG-26 class is far more complicated, more difficult to maintain, and more vulnerable to damage by shock than the system provided inherently by the series turbine.

CONCLUSION

The series turbine not only is applicable to a high speed Naval combatant ship, but also its use would be an improvement over the presently installed machinery plant.

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APPENDIX A

DERIVATION OF EQUATION OF STATE

The equation of state must cover the following range of pressures:

$p = 900$ to 1200 psia saturated

$p = 840$ to 1100 psia at 5% moisture

(1) At 1050 psia, saturated, $h = 1189.9$, $v = .4218$

(2) At 900 psia, $y = 4\%$ moisture, $h = 1168.6$, $v = .4814$

The equation of state will be in the form,

$pv = C(h-A)$.

Determination of A:

(3) $p_1v_1 = C(h_1-A)$

(4) $p_2v_2 = C(h_2-A)$

$$\frac{p_1v_1}{p_2v_2} = \frac{h_1-A}{h_2-A}$$

from (1) and (2) above,

$$\frac{1050 \times 0.4218}{900 \times 0.4814} = \frac{1189.9 - A}{1168.6 - A}$$

Solving for A,

$1189.9 - A = 1192 - 1022A$

$-2.1 = .022A$

$A = -95.4$

Determination of C

Substituting the value of A found above into equation (3),

$1050 \times 0.4218 = C \times (1168.6 + 95.4)$.

Solving for C,

$C = 0.3446$.

Substituting the value of A into equation (4),

$900 \times 0.4814 = C \times (1168.6 + 95.4).$

Solving for C,

$C = .3427.$

Let $C = .3433.$

The equation of state, then, is

$$pv = 0.3433(h + 95.4).$$

Accuracy check:

p	h	h+95.4	computed v		*steam table v		% error
Saturated							
900	1195.4	1290.8	.4924		.5006		-1.64
1200	1183.4	1278.8	.3658		.3619		1.08
At 5% moisture				$.95v_g$	$.05v_f$	v	
840	1137.5	1232.9	.5144	.5126	.001	.5136	.156
1100	1128.4	1223.8	.3906	.3801	.0011	.3812	2.46

*Keenan and Keyes, Thermodynamic Properties of steam, (New York: John Wiley and Sons, Inc, 1936).

APPENDIX B

DERIVATION OF FLOW EQUATION

For any process the relationship between pressures and specific volumes is

(1) $p_1v_1^k = p_2v_2^k$

At full power the approximate conditions are

Drum: $p_1 = 1150$, $v_1 = 0.3802$, $h_1 = 1185.6$;

S.T. outlet: $p_2 = 880$, $v_2 = 0.492$, $h_2' = 1167.2$.

Then to find the exponent, k, substitute the above values into equation (1),

$1150(.3802)^k = 880(.492)^k$,

$1.307 = (1.292)^k$.

then,

$k = 1.045$

Therefore,

(2) $pv^{1.045} = C$.

We have from Appendix A,

(3) $pv = .3433(h + 95.4)$.

Now equation (2) can be put into the following form:

(4) $\left(\dfrac{p_1}{p_2}\right)^{\frac{1}{k}-1} p_1v_1 = p_2v_2$.

That is,

$r^{\frac{k-1}{k}} p_1v_1 = p_2v_2$.

Substitute (3) for the pv terms in (4),

$r^{\frac{k-1}{k}}\left[0.3433(h_1 + 95.4)\right] = 0.3433(h_2 + 95.4)$

$$r^{\frac{k-1}{k}}(h_1 + 95.4) = h_2 + 95.4.$$

Subtracting $h_1 + 95.4$ from each side,

$$r^{\frac{k-1}{k}}(h_1 + 95.4) - (h_1 + 95.4) = h_2 - h_1$$

$$(h_1 + 95.4)(r^{\frac{k-1}{k}} - 1) = h_2 - h_1.$$

Thus

$$\Delta_1 h = (1 - r^{\frac{k-1}{k}})(h + 95.4), \quad \text{for isentropic expansion.}$$

APPENDIX C

DESIGN OF FORCED DRAFT BLOWER

PRELIMINARY DESIGN

A centrifugal flat plate fan was selected for the blower.

Design point:

120% boiler overload condition

W_A = 262,500 lb/hr (NAVSHIPS 351-0739)

RPM = 3250

η = 80%

Draft loss = 70.1 in. water (NAVSHIPS 351-0739)

$T_1 = 100^\circ F$

k = 1.4

Volume flow rate

$$Q = (262{,}500\ \frac{\text{lb air}}{\text{hr}})(14.11\ \frac{\text{ft}^3}{\text{lb}}) = 3.706{,}000\ \frac{\text{ft}^3}{\text{hr}}$$

$$= 61{,}800\ \frac{\text{ft}^3}{\text{min}}$$

Discharge pressure

$$p_2 = \frac{(70.1\ \text{in}\ H_2O)(62.4\ \frac{\text{lb}}{\text{ft}^3})}{(1728)} = 2.532\ \text{psig.}$$

$$p_2 = 14.7 + 2.5 = 17.2\ \text{psia.}$$

Head

$$H = \Delta h = c_p T\left[\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} - 1\right] = (.24\ \frac{\text{btu}}{\text{lb}\ ^\circ R})(560^\circ R)\left[\left(\frac{17.2}{14.7}\right) - 1\right](778)$$

$$= 4800\ \text{ft-lb/lb}$$

Outside diameter of vanes

$$d_2 = \frac{1300\sqrt{H}}{n\sqrt{K}}$$

K' = pressure coefficient

let $K' = .50$

$$d_2 = \frac{1300\sqrt{4800}}{3250\sqrt{.5}} = 39.3 \text{ in.}$$

let $d_2 = 39$ in.

Power required

$$FHP = \frac{HW_A}{\eta} = \frac{(4800)(262,500)}{(60)(.80)(33000)} = 796 \text{ hp}$$

Inside diameter

$$\frac{d_1}{d_2} = .78 \qquad \text{(J.T. Holm)}$$

$$d_1 = .78(39) = 30.4 \text{ in.}$$

let $d_1 = 30$ in.

Shaft diameter

$$d_s = 8 \text{ in} \qquad \text{(estimated)}$$

Inlet velocity

$$C_o = C_1 = C_{1R}$$

let $C_{1R} = 6000$ fpm

Inlet area

$$A_{shaft} = \frac{\pi}{4}\left(\frac{8}{12}\right)^2 = .37 \text{ ft}^2$$

$$A_{inlet} = \frac{\pi}{4}\left(\frac{30}{12}\right)^2 - .37 = 4.54 \text{ ft}^2$$

Vane width

$$b = \frac{Q \times 144}{C_{1R} d_1 \pi \epsilon}$$

$$\epsilon = .95$$

$$b = \frac{(61800)(144)}{(6000)(2 \text{ fans})(30)(\pi)(.95)} = 8.29 \text{ in.}$$

Radial outlet velocity

$$C_{2R} = C_{1R}\frac{d_1}{d_2} = (6000)\left(\frac{30}{39}\right) = 4620 \text{ fpm}$$

Peripheral speed (inlet)

$$u_1 = \frac{n\pi d_1}{12} = \frac{(3250)(\pi)(30)}{12} = 25500 \text{ fpm} = 425 \text{ fps}$$

Vane inlet angle

$$\beta_1 = \tan^{-1}\frac{C_{1R}}{u_1} = \tan^{-1}\frac{16000}{25500} = 13.23^\circ$$

Relative inlet velocity

$$D_1 = \frac{u_1}{\cos\beta_1} = \frac{25500}{\cos 13.23} = 26200 \text{ fpm}$$

Number of vanes

let $Z = 24$

Passage widths

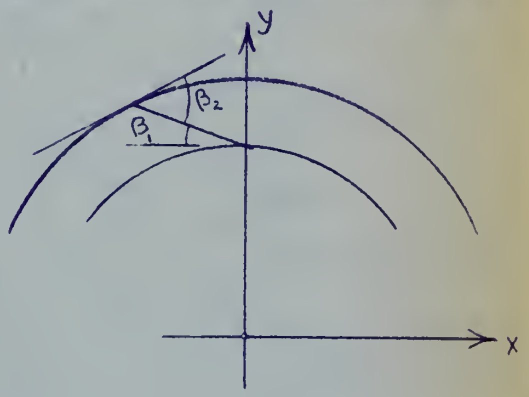
$$P_1 = \frac{d_1}{Z} = \frac{30\pi}{24} = 3.93 \text{ in.}$$

$$P_2 = \frac{d_2}{Z} = \frac{39\pi}{24} = 5.1 \text{ in.}$$

Vane outlet angle

Since the vanes are not curved, but are flat plates, the angle, β_2, can be found by analytic geometry.

The equation of the circle describing the outer circumference is



$$(1) \quad y = \sqrt{\frac{d_2}{2}^2 - x^2} = \sqrt{380 - x^2}$$

The equation of the vane is

$$y = -mx + b$$

$$m = \tan\beta_1$$

$$b = d_1/2$$

$$y = -.2355x + 15$$

The two equations are solved simultaneously to find x and y where the blade meets the outer periphery.

By simultaneous equations,

$$x = -9.24 \text{ in.}$$

$$y = -.2355(9.24) + 15 = 17.173 \text{ in.}$$

The slope of the tangent to the outer periphery is found by taking the first derivative of equation (1),

$$\frac{dy}{dx} = \frac{-2x}{2\sqrt{380-x^2}} = \frac{9.24}{\sqrt{380 - 85.2}} = .538 = \tan^{-1} 28.3^\circ.$$

Then

$$\beta_2 = 28.3 + \beta_1 = 28.3 + 13.23 = 41.53^\circ.$$

Peripheral speed (outlet)

$$u_2 = \frac{n\pi d_2}{12} = \frac{(3250)(\pi)(39)}{12} = 33200 \text{ fpm} = 552 \text{ fps}$$

Relative tangential velocity (outlet)

$$D_{2u} = \frac{C_{2R}}{\tan\beta_2} = \frac{4620}{\tan 41.53} = 5220 \text{ fpm} = 87 \text{ fps.}$$

Relative outlet velocity

$$D_2 = \frac{D_{2u}}{\cos\beta_2} = \frac{5220}{\cos 41.53} = 6980 \text{ fpm} = 116 \text{ fps.}$$

Tangential outlet velocity

$$C_{2u} = u_2 - D_2 = 552 - 87 = 465 \text{ fps.} = 27900 \text{ fpm}$$

Outlet angle

$$\alpha_2 = \tan^{-1}\frac{C_{2R}}{C_{2u}} = \tan^{-1}\frac{4620}{27900} = \tan^{-1} .1655 = 9.39^\circ.$$

Outlet velocity

$$C_2 = \frac{C_{2u}}{\cos\alpha_2} = \frac{27900}{\cos 9.39} = 28280 \text{ fpm} = 472 \text{ fps.}$$

Head developed

$$\eta_c = .50 \quad \text{(assumed)}$$

$$H = \frac{(D_1^2 - D_2^2)\eta_c + u_2^2 - u_1^2 + C_2^2\eta_c - C_1^2}{2g}$$

$$= \frac{(190800 - 13420)(.5) + 30400 - 180500 + 223000(.5) - 10000}{2(32.2)}$$

$$= 4865 \text{ ft.}$$

Virtual head

$$H_{vir} = \frac{(D_1^2 - D_2^2) + u_2^2 - u_1^2 + C_2^2 - C_1^2}{2g} = 7960 \text{ ft.}$$

Efficiency

$$\eta = \frac{H}{H_{vir}} = \frac{4865}{7960} = 61.1\ \%$$

NOTE: This design is the result of nine iterations. Only the final iteration is presented here.

FINAL DESIGN

Design point

120% boiler overload condition

W_A = 262,500 lb/hr

Q = 61,800 cfm

RPM = 3450

Draft loss = 80.2 (Design Heat Balances)

t_1 = 100°F

p_1 = 14.7 psia

k = 1.4

Discharge pressure

$$p_2 = \frac{(80.2)(62.4)}{1728} = 2.9 \text{ psig} = 17.6 \text{ psia.}$$

Head required

$$H = h = c_pT\left[\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} - 1\right] = (.24)(560)\left[\left(\frac{17.6}{14.7}\right)^{.286} -1\right] (778)$$

$$= 5535 \text{ ft-lb/lb}$$

From the preliminary design

C_1 = 6000 fpm = 100 fps = C_{1R}

d_1 = 30 in.

d_2 = 39 in.

b = 8.29 in.

C_{2R} = 76.9 in.

η_c = 50%

d_s = 8 in.

Peripheral speed (inlet)

$$u_1 = \frac{n\pi d_1}{12} = \frac{(3450)(\pi)(30)}{12} = 27080 \text{ fpm} = 451.5 \text{ fps}$$

Vane inlet angle

$$= \tan^{-1}\frac{C_{1R}}{u_1} = \tan^{-1}\frac{100}{451.1} = 12.5°.$$

Relative inlet velocity

$$D_1 = \frac{u_1}{\cos\beta_1} = \frac{451.5}{\cos 12.5} = 462.5 \text{ fps} = 27,750 \text{ fpm}$$

Vane outlet angle

As was described in the preliminary design, the equation of the circle describing the outer circumference is

$$y = \sqrt{\left(\frac{d_2}{2}\right)^2 - x^2} = \sqrt{380 - x^2}$$

The equation of the vane is

$$y = -\tan\beta_1 x + \frac{d_1}{2} = -.2217x + 15$$

The two equations are solved simultaneously

$x = -9.565$ in.

$y = 17.12$ in.

The slope of the tangent to the circle is

$$\frac{dy}{dx} = \frac{-2x}{2\sqrt{380 - x^2}} = \frac{9.565}{\sqrt{380-(9.565)^2}} = .573 = \tan 29.6^\circ$$

Then

$$\beta_2 = \frac{dy}{dx} + \beta_1 = 29.6 + 12.5 = 42.1^\circ.$$

Relative tangential velocity (outlet)

$$D_{2u} = \frac{C_{2R}}{\tan\beta_2} = \frac{76.9}{\tan 42.1} = 85.1 \text{ fps.}$$

Relative outlet velocity

$$D_2 = \frac{D_{2u}}{\cos\beta_2} = \frac{85.1}{\cos 42.1} = 114.7 \text{ fps}$$

Tangential outlet velocity

$$C_{2u} = u_2 - D_{2u} = 587 - 85.1 = 501.9 \text{ fps}$$

Outlet angle

$$\alpha_2 = \tan^{-1}\frac{C_{2R}}{C_{2u}} = \tan^{-1}\frac{76.9}{501.9} = 8.7^\circ$$

Outlet velocity

$$C_2 = \frac{C_{2u}}{\cos\alpha_2} = \frac{501.9}{\cos 8.7} = 508 \text{ fps}$$

Head developed

$$H = \frac{(214,000-13160)(.5) + (344,500 - 203,900) + 258,000(.5) + 10,000}{64.4}$$

$$= 5590 \text{ ft-lb/lb}$$

Peripheral speed (outlet)

$$u_2 = \frac{n\pi d_2}{12} = \frac{3450(\pi)(39)}{12} = 35200 \text{ fpm} = 587 \text{ fps}$$

Virtual head

$$H_{vir} = \frac{200840 + 140600 + 258000 - 10000}{64.4}$$
$$= 8990 \text{ ft-lb/lb}$$

Efficiency

$$\eta = \frac{5590}{8990} = 62.1\%$$